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SCREW PROPELLER COMPUTER

BY

GEO. R. McDERMOTT
NAVAL ARCHITECT
PROFESSOR OF NAVAL ARCHITECTURE
CORNELL UNIVERSITY

FIRST EDITION

TAYLOR & CARPENTER
ITHACA, N. Y.
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SCREW PROPELLER COMPUTER.

The Object and Basis of Its Design.

The "Computer" has been designed with the view of furnishing a means, whereby the dimensions and proportions of screw propellers—Diameter, Pitch, Surface, Revolutions, Slip and Efficiency,—can be quickly and accurately determined for any given case or set of conditions.

The formulae which forms the basis of its design and construction were deduced by the author from a rigid analysis of the results furnished by the experiments of the eminent experimentalist of the British Admiralty, Mr. R. E. Froude, and as published by him in several papers read before technical societies. The results of these experiments are recognized and used by the best informed engineers and naval architects as standard practice, and the uniform success accruing from their application in the design of propellers, for all types of vessels, naval and mercantile, has proven beyond any doubt their reliability.

The methods of application of the results of the above-mentioned experiments to every-day practice, are however, to a more or less extent, cumbersome and tedious, involving solutions of several equations, reference and cross reference to curves and mathematical tables, accompanied in many cases by a trial and error process more or less extended. For these the "Computer" is substituted; its use

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will enable the designer to arrive at a solution of propeller problems in a small fraction of the time previously taken, the results being practically identical to those obtained by the use of Froude's formulae, etc., or by the modifications of his methods proposed by several writers who have made them the basis of their writings on propulsion.

The use of the "Computer" affords also the notable advantage over that of formulae, in that the effect on the dimensions, of changes in one or more of the related quantities is clearly shown, and may be obtained by mere inspection, thereby enabling the designer to come more quickly to a final and accurate decision as to the most suitable dimensions of propeller for any given case.

In the analysis of the performance of propellers and ships the "Computer" is also of great value, giving at once the results sought, and which if calculated by the use of formulae, would involve a large expenditure of time and trouble.

The action of a screw propeller in developing a thrust is relatively complex, and although a full discussion of the problem or of the many theories which have been advanced towards a solution is beyond the scope of this work, in order that an intelligent use of the "Computer" be made, it has been deemed desirable to give a brief outline, in the simplest possible form, of the leading features of screw propulsion.

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GENERAL DISCUSSION OF SCREW PROPULSION.

Resistance of the Ship.

When a ship is towed through the water at a given speed V, it meets with the following resistance:

Resistance of the Water.—(1) Frictional, (2) Wave-making, (3) Stream-line, (4) Eddy-making, the sum of the last three being known as the "residuary resistance" and the total as the net resistance, r.

Resistance of the Air.—In addition to the resistance from the water, the vessel also encounters resistance from the air on the parts of hull and upper works above the water, rigging, etc.

Thrust.

The thrust necessary to overcome the resistance to the forward motion of the vessel is developed, by the propeller projecting a column of water with a certain velocity in a direction opposite to that of the ship; the reaction from the sternward momentum given to the water —per unit of time—constituting the thrust which transmitted through the shafting to the thrust bearing, drives the vessel ahead.

Rotation of the Race.—A portion of the total acceleration given to the column of water is gradually imparted to it forward of the propeller, the back of the blades exerting a suction, and in passing through the propeller it receives the final sternward acceleration and at

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the same time is given a rotary motion which is accompanied by centrifugal force causing a diminution of pressure from the center to the outer limits of the column. The effect of this rotation is to cause a larger proportion of the final sternward acceleration to be imparted to the water before passing through the propeller: it is generally accepted, however, that this effect may without material error be ignored, except with high slip and pitch ratios.

Cavitation.—There is a limit, however, in the process of developing the thrust, to the amount of suction or pull that can be exerted on the water forward of the propeller; if this limit is reached a cavity is formed between the water and the blades of the propeller thereby depriving the screw of a portion of its natural thrust-producing power. Assuming that the tips of the propeller blades are a little below the surface of the water a cavity will form, if the tension or pull on the water is or exceeds 15 pounds per square inch, provided that the space forms a perfect vacuum, but as the cavity would probably contain water-vapor and air given off by the water, "Cavitation" may be expected to occur at some lower value.

The phenomenon of cavitation has received attention at the hands of several experimentalists, notably Mr. Sidney W. Barnaby, who, from experiments made on the British torpedo boat "Daring," found that there was a definite thrust per square inch of blade surface at which cavitation commenced. With the tips of the blades 11 inches below the surface of the water, he found that the limiting value of the thrust was 11¼ pounds per square inch, with an increase of ⅝ of a pound for each additional foot of immersion.

Mr. Charles Parsons, designer of the "Turbinia" and "Cobra" has, from independent experiments, corroborated Barnaby's figures.

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Happily for the designer, however, it is only in very high speed vessels—torpedo boats, destroyers, etc.—that the limits are reached where trouble may be encountered from cavitation. In these vessels care should be taken to provide such amount of surface to the propellers that the thrust per square inch does not exceed 11 to 12 pounds, otherwise a decreased efficiency of propeller will result.

It is to be noted that the above values are referred to the projected surface of the blades.

Reactions Between Propeller and Ship.

The Wake.—When a vessel is propelled instead of being towed through the water, certain reactions take place between the propeller and the ship, which it will be well at this point to describe.

Considering first, the effect of the presence of the ship on the propeller.

When the vessel is propelled through the water at a given speed V it is accompanied by a surrounding zone or belt of water, which has a forward motion impressed upon the particles of water within it, caused chiefly by the friction between the hull of the ship and the particles immediately in contact; the forward velocity of this frictional zone increases as the stern is approached, at the same time decreasing at points farther and farther removed from the hull of the ship in an athwartship direction, and as may be readily conceived, the longer the vessel the greater, relatively, will be the forward velocity of the "frictional zone" wake.

In addition to the "frictional zone" wake, the stream line motion, and the waves propagated by the advance of the ship contribute a forward velocity to the water adjacent to and surrounding the vessel,

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forming what may be termed the "residuary" wake; the forward velocity of the residuary wake, at any given speed of the ship, varying with the character of the form and the proportions of the vessel.

The combined wake, made up of the "frictional zone" and "residuary" constituents, is more or less turbulent in character, with varying velocities throughout its cross section, the maximum being found at the fuller portions of the stern of the ship and near the water-line, diminishing in the vicinity of the finer portions of the deadwood near the keel, and similarly to that of the "frictional zone" component, at successive points distant from the vessel athwartships.

From a number of experiments, made by Froude and others it has been determined, that although the motion of water forming the wake is complex and variable, so far as its influence on the action of screw propellers, it may for all practical purposes be considered, as a current having a uniform velocity throughout the athwartship field in which the propeller operates.

Speed of the Wake—$w\ V$. The speed of the wake may be conveniently expressed as a fraction of the speed of the ship; denoting this fraction or "wake factor" as it is more commonly called by w we have speed of wake $= w\ V$. For values of w see Table 1.

Speed of Water in which the Propeller works—V_1. This is equal to the speed of ship less the speed of wake or $V - wV = V(1-w) = V_1$.

Influence of the Wake on Propeller.—It will be readily seen, from a consideration of the foregoing, that a propeller working in the water forming the wake, derives therefrom an increased thrust, for the same number of revolutions and speed of advance, as compared with that which would be obtained were it working in water undis-

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turbed by the presence of the ship. It should be noted that as the presence of the wake is due to the motion of the ship, the gain in thrust obtained therefrom represents simply a return of a portion of the energy expended by the ship on its formation, the engines being the original source of supply.

The gain in thrust coming from the wake, differs in amount in the case of single screws from that of twin screws; in single screw ships, the propellor being situated at the middle plane of the ship is immersed in and subjected to the combined frictional and residuary wakes, whereas the twin screw propeller as ordinarily fitted—with the tips of the blades well clear of the hull—operates only in the residuary wake which has a smaller mean forward velocity than the two combined.

The farther that either single or twin screws are removed aft of the stern of the ship, the less will their performance be influenced by the wake, and consequently, the smaller will be the gain in thrust.

Augmentation of Resistance or Thrust deduction.—Considering now the effect of the presence of the propeller on the performance of the ship.

A screw propeller turning at a given number of revolutions, due to a portion of the final acceleration of the water projected sternward being imparted to it forward of the propeller, exerts a suction upon the water lying between it and the vessel, producing a defect of pressure, thereby interfering with the natural closing in of the stream lines around the afterpart of the ship, causing an increase in the resistance of the vessel, and necessitating the development of an additional thrust t, from the propeller to overcome it.

Viewed from the standpoint of the ship this increase may be termed an "augmentation of resistance" but if from the standpoint of

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the propeller it may be more properly stated as a "thrust deduction" representing as it does, the withdrawal of a portion of the thrust production of the propeller which would otherwise have been usefully expended in overcoming the net-resistance of the ship, or, in other words, a loss in thrust.

Similarly to that of the wake, the thrust deduction differs in amount in single screw steamers as compared with twin screws; the single screw situated at the middle plane and in close proximity to the afterbody of the hull deprives the vessel of a greater intensity of pressure—coming from the stream line motion in combination with the frictional and residuary wakes—thereby causing a greater loss, than would be the case in twin screw ships the propellers of which being farther removed from the hull and acting in the residuary wake region only, interferes to a smaller extent with the stream line pressures creating less "augment of resistance" and consequently smaller "thrust deduction."

The farther that either single or twin screw propellers are removed aft of the stern the smaller will be the loss in thrust.

Interdependence of the reactions between the ship and Propeller.—As may be readily seen on comparing the mutual reactions between ship and propeller, they are largely interdependent; the conditions that are productive of an increase in the gain from the wake, are favorable towards an increased loss from the thrust deduction, the one tending to offset the other in the effect on the performance of the propeller and in their relation to the complete phenomena of propulsive efficiency.

Wake gain factor—$(1-w)$. The gain in thrust coming from the wake is measured, and may be conveniently expressed, by what we

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may term the "wake gain factor," *i. e.*, the ratio of the speed of the water in which the propeller works V_1 to the speed of the ship V. This ratio may be stated as follows: $\frac{V_1}{V} = \frac{V(1-w)}{V} = 1-w$.

Thrust deduction factor—$(1+t)$. The loss due to the thrust deduction, is expressed by what is termed the "thrust deduction factor," or, ratio of the thrust T necessary to overcome the net-resistance plus the augmentation, to the thrust T_1 required to overcome the net-resistance only, or $\frac{T}{T_1} = 1+t$. t representing the increase in thrust.

Coefficient of reactions—C_r. Whatever efficiency a propeller might have working by itself in water undisturbed by the presence of the ship, that efficiency would be effected by being brought into conjunction with a ship on account of the mutual reactions referred to. This effect may be expressed by the product, of the wake gain factor by the thrust deduction factor or $(1-w)(1+t) = C_r$, this being termed by the author the "Coefficient of reactions" to distinguish it from Froude's "hull efficiency factor," the one being the reciprocal of the other.

As the result of his experiments Froude has stated that, the value of the "Coefficient of reactions" is roughly equal to unity, but from an examination of data on this point, the writer is inclined to the opinion that for twin screws as ordinarily fitted the value would be found more nearly between 1.02 and 1.04 showing a slight advantage in favor of single screws, having a value of 1.0 as stated by Froude. No great error would result, however, if the value be taken as 1.0 for all cases.

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Attention must be drawn to the great probabilities, that for single screw vessels with thick sternposts, such as wooden composite and sheathed ships, and for small wooden launches and yachts where care is not taken to taper the after edge of body post and the forward edge of rudder post, that the value of C_r will be greater; insufficiency of data prevents any definite statement being made, but from the information at hand the indications are that it might be as high as 1.1 to 1.2.

Horse Powers.

Effective horse power—EHP. We have on a previous page stated the several forms of resistance encountered by a vessel when towed through the water at a given speed V, the summation being the net-resistance, requiring a thrust T_1 to be developed by the propeller to overcome it. The product of the thrust by the speed of ship is called the effective horse power, or, $T_1 V = EHP$.

Thrust horse power—(apparent propeller horse power)—*THP*. We have found, that due to the presence of the propeller the net-resistance r is increased by the amount of the augmentation, requiring an increased thrust T to overcome it. The product of the thrust T by the speed, is known as the thrust horse power, or $TV = THP$. It may be noted that $EHP(1+t) = THP$. *THP* for the reasons given below may be called the "apparent" propeller horse power.

Propeller Horse power—PHP. It would seem at first sight that as the thrust horse power *THP* comes from the propeller, it might equally be called the propeller horse power, but that it is only an apparent propeller horse power is evident when it is remembered, that a portion of the total thrust T is gained from the wake, and only

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that remaining can be credited to the propeller as its own natural production.

The gain from the wake may be eliminated by means of the wake gain factor $(1—w)$ as follows:—

We have,

T = total thrust overcoming net-resistance plus augment.

$TV = THP$ = total thrust horse power from propeller.

$T(1—w)$ = total thrust minus gain from wake.

$TV(1—w) = THP(1—w) = PHP$ or true propeller horse power.

The following relations may also be shown:

$EHP(1+t) = THP$, hence $EHP(1+t)(1—w) = PHP$

but $(1+t)(1—w) = C_r$, therefore $EHP \times C_r = PHP$

In cases where the "coefficients of reactions" $C_r = 1.0$ *i. e.*, where the gain from wake offsets loss in thrust deduction then $EHP = PHP$.

Delivered horse power—DHP. We have in the propeller horse power stated above, the amount of power utilized by the propeller in imparting a sternward velocity to the column of water; in addition to this a certain amount of power is absorbed by the propeller in overcoming its own frictional and edgeways resistance, in revolving through the water, as also losses due to rotation, etc., the sum total representing the power which has to be delivered to the propeller by the engines, the amount depending upon the true efficiency of the propeller. Denoting the efficiency of the propeller by e we have

$$\text{Delivered horse power} = \frac{PHP}{e} = DHP.$$

Indicated Horse Power—IHP. In the process of developing and delivering the power required by the propeller, a certain additional power is absorbed, in overcoming the initial and load

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friction of the engines and shaft bearings, and in the working of any attached pumps.

Initial friction.—Represents the friction coming from tightness of pistons, shaft and other bearings, glands, valves, etc.; it is generally believed to be constant, and consequently, that the power absorbed in overcoming it varies directly with the number of revolutions.

Owing to the difficulties lying in the way of experimental determination of the amount of initial friction, very little data is available, but from such experiments which have been made, the value seems to lie between 1.6 and 3.0 pounds per square inch of low pressure piston area.

With a mean effective pressure—m.e.p.—referred to L. P. Cylinders of 33 pounds, this would represent from 5 to 9 per cent. of the maximum *IHP*, varying directly as the revolutions at lower powers.

Load friction.—The information regarding the value of the power absorbed by the friction of the load is still more uncertain than that concerning the initial friction, but, there is good reason for believing that at full power it equals that absorbed by the initial friction, *i.e.*, 5 to 9%; varying directly as the *IHP*, at lower powers.

Power absorbed by attached pumps.—This will depend on the number and design of the pumps worked directly off the main engines the amount of power required may be stated as follows:

	Per cent of *IHP*.
Circulating pumps from	.75 to 1.5
Air " "	1.0 to 1.5
Feed " "	.5 to .6
Bilge " "	.5 to 1.0

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The power absorbed by the initial and load friction increases with the number of cylinders and bearings, so that for vessels having the greater length of shafting, etc., the value chosen should approach the higher of those given. The figures given are for modern 3 cylinder engines.

It is natural to suppose that the friction of new engines will be decreased as the bearing surfaces are worn down and become self-adjusted, on the other hand careless adjustment and lining up, when fitted on board (unfortunately of frequent occurrence) as also the excessive working of the ship would lead to large losses from friction.

Assuming that the air pumps alone are attached to the main engines it may be seen from the above that the total friction of the average engine may be stated as lying between 12 and 20% of the maximum *IHP*.

The sum of the *DHP* and friction horse power gives the total horse power to be developed in the cylinders or *IHP*.

Efficiencies.

The term efficiency as applied to either the propeller or engine, singly or in conjunction, is simply the ratio of the power realized to the power expended.

Efficiency of the Engine.—e_m. The ratio of the power delivered to the propeller, to that developed in the cylinders, denotes the efficiency of the engine or $DHP \div IHP = e_m$.

From the values given in the discussion of the losses due to engine friction, it follows that the value of e_m at maximum power may be stated as between .88 and .80 for engines of good design and construction. It is possible that in small launch engines, and for larger

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engines where exceptional care is taken in their design and fitting on board, the value of e_m may approach .9 to .92.

Apparent Efficiency of Propellers—e_a. The ratio of the thrust horse power, to that delivered to the propeller by the engine expresses only, for the reasons already given, the apparent efficiency of propeller $= THP \div DHP = e_a$.

True Efficiency of Propellers—e. As has been already shown, the thrust horse power used in the preceding ratio includes not only the utilization of the power delivered to the propeller by the engines, but also that gained from the wake. The true efficiency, considered simply as a propeller, should express its capability of utilizing the power delivered to it by the engine only, *i.e.*, the ratio of the useful work to that expended; this may be obtained from the ratio of the propeller horse power to that delivered, or $PHP \div DHP = e$.

If the apparent efficiency e_a is known the real efficiency may be obtained, by multiplying it by the wake gain factor, or $e_a(1-w) = e$.

The experiments of Froude, Barnaby and Blechynden, with propellers having various proportions, shape of blades, surface, pitch and slip ratios, have shown conclusively, that the efficiency e of a well designed propeller, operating under the most favorable conditions, does not exceed .7; those of Froude on screws having elliptical form of blades (the shape adopted as "standard" in this work) showed a maximum value of $e = .69$.

Propulsive Coefficient—pc. The ratio of the power realized —effective horse power—to the power consumed—indicated horse power, is a final measure of the net-total efficiency of propulsion, and is expressed by the well known "propulsive coefficient" $EHP \div IHP = pc$.

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From the relations discussed in preceding pages it may be readily seen that the propulsive coefficient is the product of the values expressing the true efficiency of the propeller by the efficiency of the engines, divided by the coefficient of reactions, or, $pc = \frac{e \times e_m}{C_r}$

The following example will illustrate the derivation of the various coefficients which have been discussed, the distribution of horse power; the amount and source of the losses, in passing from the engine to the ship.

Let

Engine efficiency.....e_m = .8	*Thrust loss factor*.....$(1+t)$ = 1.3
Efficiency of propeller..... e = .69	*Coefficient of reactions* $(1-w)\times(1+t) = C_r$ = 1.04
Wake gain factor.....$(1-w)$ = .8	*Propulsive coeff*.....pc = .5308

Distribution of power.

IHP	=	10000
$IHP \times e_m$	= 10000 × .8 = DHP =	8000
$DHP \times e$	= 8000 × .69 = PHP =	5520
$PHP \div (1-w)$	= 5520 ÷ .8 = THP =	6900
$THP \div (1+t)$	= 6900 ÷ 1.3 = EHP =	5308

Losses.

Friction of engines	$IHP - DHP$ =	2000
Friction, etc., of propeller	$DHP - PHP$ =	2480
Reactions	$PHP - EHP$ =	212
Total losses	=	4692
Power expended	IHP =	10000
" *realized*	EHP =	5308

Propulsive Coefficient = $pc = \frac{5308}{10000} = .5308$

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We have in the preceding pages discussed the leading features of screw propulsion; the discussion has necessarily been of the briefest character, but it is believed by the writer to be sufficient for the purpose intended, that of placing before the reader in a plain and concise form, the various influences which have to be taken account of in determining the dimensions and proportions of a screw best suited for a given ship.

The rule-of-thumb methods which were used—and in less informed practice are still in use—before Froude's experimental investigations furnished the designer with the necessary information regarding the various elements involved, consisted in taking the propeller of as nearly a similar vessel as lay within the designer's knowledge and which had done well on trial or in service, proportioning therefrom, a propeller for the new vessel, by means of more or less incomplete and crude formulae, which may be found even at the present day in what are quoted as standard engineering manuals; the use of these methods and formulae requiring the exercise of considerable judgment, and affording large opportunities for the exhibition of results partaking of the hit-and-miss order, arising chiefly from the fact, that their use affords no means of taking separate account of the individual merits and characteristics of the propellers adopted as types, or of the mutual reactions between them and the vessels to which they were fitted.

Froude's experiments were undertaken and made with models of propellers and ships—separately and in conjunction—with the following objects kept clearly in view. To obtain information regarding the thrust and efficiency of screws having different pitch slip and surface ratios, also the influence of the ship on the propeller and that of the

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propeller on the ship, and the adjustment and verification of the laws of comparison between propellers of similar type but dissimilar dimensions. These objects were amply fulfilled, and it is now fully and definitely established, that, for a given screw each selected pitch ratio has a certain slip ratio with which it must work to attain maximum efficiency, and that with larger or smaller slip ratios a reduction in efficiency will inevitably result.

Furnished with this information, and if for a given ship the *EHP* is known for the required speed, also the number of revolutions suitable for the seleted type of engine, the diameter pitch and surface of propeller could be readily computed, which would yield the required thrust, while working with the slip corresponding to the maximum, or such other efficiency, which the conditions surrounding the design as a whole would permit. The average designer, however, possesses only a knowledge of the *IHP* required for the given speed, and the number of revolutions necessary to develop the power; nevertheless, with values selected from the data given by the author, of the wake, propulsive, and other coefficients, appropriate to the ship and selected type of engine, the propeller having the most suitable dimensions can be readily obtained by means of the "Computer."

In comparing results obtained by the "Computer" with those gotten by the use of Froude's formulae, it is to be remembered that in putting his data into shape for publication he framed it so as to include an assumed wake of 10% of the speed of the water V_1 in which the propeller works. The author, as may be observed, refers the speed of wake to the speed of ship, and in designing the "Computer" has eliminated the 10% wake assumption, leaving it to the user to give effect to such wake as may be appropriate to any given case, in the

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manner described in the directions for operating. Froude also selected a certain point in the efficiency field as the maximum—corresponding to what is known in the nomenclature adopted by him, as the "9 abscissa value." The author for several reasons has selected the "10 abscissa value" instead; the selection is based largely on the fact, that extended experience has shown that better results are thereby attained, on all classes of vessels. Directions are furnished, however, for those who may prefer to use the "9 abscissa value" in the employment of the "Computer" for determining the dimensions of propellers, in which case, the dimensions obtained will be practically identical with those derived by the use of Froude's original data.

We will now take up the discussion in detail, of the several quantities involved in the determination of the dimensions of screws by by means of the "Computer."

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QUANTITIES INVOLVED IN DETERMINING THE DIMENSIONS OF PROPELLERS.

We have seen in the preceding discussion on screw propulsion, that the problem before us is the determination of the dimensions of one or more propellers capable of developing the necessary thrust, with the best possible efficiency under such conditions as may be imposed by the nature of the service and form characteristics of the vessel.

In proceeding to obtain these particulars, the following details, related to the propeller and the ship, which have a direct influence on the final selection of the dimensions of the screw, have to be considered and the values of the quantities involved, definitely decided upon or estimated for any given case.

Propeller.	*Ship.*
Number of propellers.	*Speed V in knots.*
Number of blades.	*Indicated horse power.*
Shape of blades.	*Wake percentage value.*
Surface ratio.	*Propulsive coefficient.*

A discussion of these taken in the above order will assist in deciding upon their values for any given ship.

Number of Propellers.—The question whether a vessel should be fitted with one, two, or more screws, is largely one in which the conditions of the service for which the vessel is to be designed may exercise a considerable influence.

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In the mercantile marine, passenger vessels in both ocean and coastwise routes have as a rule relatively high speeds, requiring large powers, which may call for a subdivision on account of the limit to the amount of power which may be considered possible or desirable to transmit through one shaft; limitations of draft may also place a limit on the diameter of propeller, keeping in view the necessity of a proper immersion of the blades, thereby requiring the fitting of more than one, of the limited size, to obtain the required thrust.

Duplication of machinery and propellers in these vessels miminizes the risk of total disablement, and on account of the smaller sizes of working parts and shafting, facilitates repairs. Weight economy may also be obtained by the adoption of more than one screw, as higher piston speeds may be deemed more practicable with smaller engines under a given set of conditions. First cost, running expenses, and space occupied, have, however, to be taken into account, these being usually greater in multiple screw vessels.

Cargo steamers are usually fitted with single screws, but cases may occur where the conditions of service—draft, etc.,—may render the adoption of twin screws preferable. In large ocean freighters, increased cargo space may often be obtained by fitting twin screws, their machinery taking up athwartship space that could not be utilized for the stowage of cargo, and although these vessels carry a comparatively small number of passengers, the vessels themselves and the cargo carried, as they increase in size, become more and more valuable, rendering it a commercial necessity to safeguard the certainty of their reaching port, and this may take the form of duplication of machinery and propellers.

War vessels of all types, in addition to the above, have other important requirements to be fulfilled, such as great manoeuvering power,

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reduction of exposure of the machinery to projectiles, and consequent smaller chance of total disablement in action, and the securing of the best economy under steam at the several speeds of service—full power, cruising, squadron, etc.,—the consideration of all these requirements render the duplication of machinery of vital importance in the larger number of naval vessels.

High speed vessels of the "torpedo" class may require duplication of propellers to avoid the evil effects of "cavitation;" the point where this occurs has been already discussed.

Triple screws, in high speed warships of larger size have come into prominence, largely through the indomitable energy of Admiral Melville, U. S. N., and in spite of opposing opinions seem to be gradually working their way into favor with warship designers, and further increases in speed and power will probably call for their adoption in mercantile as well as in naval ships.

Number of Blades.—If the blades of a propeller were infinitely thin it is quite possible that a screw having, say, six or eight blades, would be at least as efficient as one having a smaller number, but, as the blades—with the same diameter, shape and total surface—require to be of a certain thickness to give them sufficient strength, the efficiency of the screw would probably be lowered, on account of the mutual interference of the large number of blades and greater edgeways resistance; apart from the mechanical efficiency, the increased cost compared with that of propellers having a smaller number of blades, is probably one of the reasons why they are not found in current practice.

While it is a well established fact that equal propeller efficiency may be expected with two, three or four bladed propellers having properly adjusted dimensions and proportions, it will be well to consider

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in detail the application of these, to both single and twin screws vessels. Although equally efficient in smooth water, the performance of two bladed screws is more seriously affected by the pitching and yawing of the ship in a rough sea, setting up larger vibratory and other strains, than either three or four bladed propellers under similar conditions of open sea work, and should one of the two blades be broken off, the engines would be subjected to unequal straining; the same general reasoning applies to three bladed propellers, when fitted to single screw ships, and as a consequence these vessels are invariably fitted with four bladed propellers.

The propellers of twin screw ships, owing to the subdivision of power, have relatively smaller diameters, and consequently deeper immersion, and are not influenced, to so large an extent, by rough weather conditions as that of the single screw; therefore, the reasons which render it advisable to fit four bladed propellers to single screw vessels, do not apply so directly in the case of twin screws; nor in view of what we have seen to be the bad sea-going qualities of two bladed screws, would it be advisable to fit them to the latter vessels. Taking everything into consideration, the three bladed propeller would seem to be the most suitable for twin screw ships.

Current practice corroborates these opinions, it being almost the universal custom, to fit three bladed and four bladed propellers to twin and single screw ships respectively, in both the naval and mercantile marines. Should it be necessary, however, in twin screw vessels, to obtain smaller diameter of propeller, on account of shallow draft, four blades may be employed without hesitation, provided, the width of blades is not excessive.

Shape of Blade.—The forms of blades which have been tried, since the application of the screw to the propulsion of vessels, are al-

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most infinite in number and variety, beginning with the "fantail" having greatest width at the tips, passing to rectangular, and finally the pear shaped, with the greatest width at or in close proximity to the hub or boss, all of these having the corners more or less rounded and the blades twisted and bent according to the ideas and fancies of the respective makers.

Experiments have shown, that the propellers having the blade area distributed nearest the tips develop the greater thrust per unit of area—the old fantail blade being representative of this class. Blechynden's experiments showed the following relations between the thrust of different shapes. Elliptical = 1.0; Griffith's .97; rectangular 1.115; fantail 1.17. But, it is to be borne in mind, that, although the fantail blade may develop greater proportionate thrust, it is accompanied with an increase in friction, due to the distribution of the area nearer the tips where the circumferential velocity is greater, and as the blades have to be made thicker to withstand the greater bending moment an increased edgeways resistance must also result, the total increase requiring a proportionately greater delivered horse power *DHP*, and a resultant lower efficiency, than would be obtained with blades of less abnormal shape. Going to the other extreme, when the greater proportion of the area is situated close to the boss, inefficiency results, from the fact, that the excess of area nearer the center contributes a very small increase of thrust, while it expends a proportionately greater power in "churning" the water. The rectangular form of blade possesses the disadvantages, to a modified extent, of the two extreme types just discussed without affording any advantages in return.

Forty years ago Mr. Griffiths, who received his first lessons in propulsion from Capt. Ericsson, brought forward what was at that date

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a new form of blade; the original "Griffiths" blade was to a certain extent pear shaped, the widest part being situated about .4 of the diameter from the center of the propeller, the tips of the blades were also inclined forward about .4 of the diameter. This propeller gave such excellent results that it very quickly found favor in both naval and mercantile circles.

At the present day, what may be looked upon as the standard is known as the "modified Griffith's," having the blades more or less elliptical, and from the experience gained it is believed to be capable of yielding results as satisfactory as any other form, and certainly superior to the fantail, rectangular, or abnormal pear shaped blade.

It is readily seen why the elliptical shape is superior, as the portions of the area, which is detrimental at the tips, due to the greater friction: as also that near the hub, is removed to about midway between, delivering their proportion of the thrust, without expending unnecessary power in overcoming needlessly great friction or in "churning" the water.

The elliptical form of blade is becoming, more and more, to be universally adopted in naval and mercantile vessels under the name already given—"modified Griffiths." Slight departures from a true ellipse may be from time to time observed in standard practice, but while these may do no harm, it has never been made clear what special benefit is derived. Oftentimes, changes in shape are made on a propeller which has been already tried, and, the improved results obtained after the changes, are credited to the supposed better shape, whereas, they are probably due to the cutting away what was really an excess of area, in the sense, that it contributed little or nothing to the general result, while it consumed power, which in the modified form,

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was expended in a farther acceleration of the column of water driven sternward, with a corresponding increase in the speed of the ship.

In carrying out his experiments on screw propellers, Froude adopted the elliptical form of blade; the major diameter extending from the tip to the center, the inner end of the ellipse being cut off by the boss; the minor diameter or greatest width of blade, situated at the center of the radius of screw, was equal to .2 of the diameter of propeller. The helicoidal or expanded blade area, deducting that cut off by the hub, was equal to .09 of disc area swept by the blade; so that his two, three and four bladed propellers had surface ratios of .18, .27, and .36 respectively.

Departures may be made from above width of blade ratio, providing, that they do not exceed the limits where, inordinate widths would cause interference and consequent diminution of the efficiency; these may be stated as about .275 and .375 of the diameter of 4 and 3 bladed propellers respectively. If for any reason greater surface is required than these width values would afford, an increased diameter should be adopted, rather than an increase in width.

Surface ratio—*b*. In dealing with the surface of propellers, it is convenient to refer to it as a fraction of the disc area, or $B \div A = b$, where, B is expanded area of all the blades, A, area of circle swept by the tips of the blades, and b, surface ratio.

In selecting an appropriate surface ratio for a proposed propeller, due consideration must be given to the type of vessel and the nature of the service for which she is intended; and, owing to the interdependence of all the dimension characteristics of propellers—diameter, pitch, revolutions and slip—these should also be taken into account, in determining the most suitable surface ratio.

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In both single and twin screw ships, having full bluff stern lines, especially if accompanied with broad beam, the feed of water to the propeller is to a greater extent interfered with, and necessitating relatively larger diameter of propeller than those of finer vessels, reaching out into the less disturbed water; this result may be obtained by the adoption of a relatively small surface ratio; for example, assuming an average lined vessel having a propeller of normal diameter with appropriate pitch ratio, surface ratio and revolutions : relatively, for maximum efficiency, a bluff full ship should have a propeller according to some one of the following modifications: smaller surface ratio, finer pitch ratio, same revolutions, and larger diameter; or, smaller surface ratio, same pitch ratio, smaller number of revolutions, and larger diameter; if the surface ratio were the same in both vessels, the bluff ship to obtain larger diameter, would require higher pitch ratio and smaller number of revolutions, but experience with full cargo vessels has demonstrated that high pitch ratio is not suitable, for the reasons given under "pitch ratio" discussion, page (40) this being the case, smaller surface ratio is evidently the better choice to obtain an advantageous diameter; nevertheless, the selected type of engine may demand a certain number of revolutions, forcing the adoption of relatively less efficient proportions and dimensions of propeller. Table 5 shows the relations which exist between the characteristics of propellers and the effect of the change of one of the values—decrease or increase—on the value of the others; a reference to this table will aid the designer in deciding, the best course to pursue, in any given case where difficulties are encountered on account of opposing conditions.

Table 2 gives the values of the surface ratios of propellers that are recommended for adoption, in vessels having form characteristics as stated therein. These values represent deductions, not only from the

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experimental data of Froude, Blechynden and others, but also from the experience gained from the best standard practice; they may, however, be departed from slightly, to suit abnormal forms, limitations of diameter or number of revolutions suitable for the type of engine selected, but in no case should the surface ratio exceed .55 as trustworthy experiments have shown, a very rapid decline in efficiency at about that value, and no gain in thrust need be expected, when the value is .6 or above. For these reasons, the scale of surface ratio on the "Computer" does not extend beyond .5.

Speed—V. As the average vessel and her machinery are designed to work with maximum efficiency at the highest speed, it is proper that the propeller should be designed for the same speed; and, as is customary in estimating the horse power required for a given vessel, the maximum mean-speed on trial should be taken as the basis in determining the dimensions of the screw; the efficiency of its performance will not be appreciably affected by the difference which ordinarily exists, between the speed on trial, and that usually attained on service, under ordinary seagoing conditions.

When the conditions of service are such, as to render it desirable to have the best all round efficiency of propulsion at some lower speed, such as the cruising speed of warships, etc., the lower speed, or some speed between it and the maximum—depending on the actual requirements—should be taken in computing the dimensions of the propeller.

As tugboats and their machinery should be designed to attain maximum efficiency while towing, the speed which is estimated the boat will maintain while towing should be used in designing the propeller.

Likewise in case of vessels engaged in trades in which continuous

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head winds and sea prevail, resulting in a considerable reduction from the speed on trial while developing full power, the expected service speed should be taken as the basis.

Horse Power—IHP. The indicated horse power corresponding to the speed is to be used, the necessary reduction to propeller horse power being taken care of in the design of the "Computer," and by the use of appropriate values of the propulsive coefficients, etc., as explained later.

The maximum indicated horse power, corresponding to the highest mean-speed, should be used in every case, with the exception of those vessels—warships, etc.,—that may be designed to have maximum efficiency at some lower speed; in these cases the power appropriate to the lower speed should be used.

In vessels with multiple screws, twin, etc., the indicated horse power of one set of engines only, is to be taken in computing the dimensions of each of the propellers.

Wake Percentage—$100\,w$. For the reasons already stated in the general discussion of screw propulsion, the propeller should be designed with reference to the speed of water in which it works or V_1, this is given effect to in using the "Computer" by the employment of the "wake percentage" ($= 100\,w$).

As we have seen, in the discussion of the origin and constituents of the wake, the propeller of the single screw ship works in the combined frictional and residuary wake, the velocity of which is influenced by the length, form and proportions of the ship, whereas the twin screw operates in the residuary wake only, the speed of which is affected by the form and proportions of the vessel; consequently, the values of the wake percentages are different in the two cases.

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Accurate values of the wake % for any type of twin and single screw vessels are given in Table 1; the mean value of the wake is also affected by variations in the proportions of draft to breadth of ship and in framing the table a fair average has been assumed.

The values for twin screws are obtained direct from the division of the table headed—Twin Screws—, while those for single screws are obtained, by selecting, from the Twin screw division, the value appropriate to the form and proportions of the vessel, adding to this, the value selected from the division of the table headed—Single screw additions —appropriate to the length of ship.

Take as an example:—Required wake % for vessel. Length = 460, Breadth = 57, Draft = 23, prismatic coefficient = .66: values to be given, as a twin screw; and as a single screw.

	Twin Screw	*Single Screw*	
		Unsheathed	*Sheathed*
From Twin screw division of table	6.4	6.4	6.4
" *Single* " " "		15.7	26.8
Wake %	6.4	22.1	33.2

In triple screw vessels, the wake % values for the center and side propellers are to be obtained, as for single and twin screws respectively.

Propulsive Coefficient—*pc.* As already stated on a previous page the propulsive coefficient is the ratio of the power realized to that expended or $EHP \div IHP = pc$.

Table 2 gives what may be regarded as fair representative values of *pc* for different types of vessels, from which may be selected, the value appropriate to any given case, for use with the "Computer;" the

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values given are the result of a careful examination of data of a number of vessels, the *IHP* of which were accurately obtained on measured mile trials, and the *EHP* by tank experiments with their models.

It has also been shown that the propulsive coefficient is equal to the product of the efficiencies of machinery and propeller divided by the coefficient of reactions or $pc = \frac{e_m \times e}{C_r}$; if for any given case, accurate values can be assigned to the several efficiencies, the value of *pc* may be obtained from the above equation. In selecting the value of *pc* direct from Table 2, due consideration should also be given to the probable values of the different efficiencies involved, these have been stated with as careful an analytical subdivision, under their several heads, as present experience permitted, and in giving this consideration, it is desirable that the following related facts are kept clearly in view.

e_m.— The difficulties attending the determination of the initial friction of the engine, and the insecure data of the friction coming from the load, renders it impossible to make definite statements as to the exact value of e_m for the varied types of engines.

e.— While the maximum value, $e = .69$ determined by Froude's experiments, is reliable as applied to model propellers, there remains the uncertainty, as to the exact nature of the modifications which may affect the thrust and co-related quantities when we pass from the model to the full sized propeller—the amount of correction of the friction of the blades may be cited as one of the quantities upon which there is very indefinite information—these modifications may cause the possible maximum value of *e* of actual propellers to be some other figure

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than .69, but whether this is so or not, we are not directly concerned; the best course to pursue, is to design for maximum efficiency irrespective of its absolute value; should the conditions not permit this, other values—1%, 2%, 3%.....less, with either greater or less slip than that related to the maximum efficiency (see efficiency scales of "Computer")—will have to be selected in computing the most suitable dimensions of screw.

C_r— Although this is usually taken as equal to unity for all cases, it should be borne in mind, that abnormally full afterbody, the locating of propellers too close to the stern, and abnormally high pitch and slip ratios, increases considerably the thrust deduction, or, augmentation of resistance, and likewise the value of C_r.

Fortunately the design of a propeller for a given ship does not call for a precise knowledge of the absolute values of these efficiencies, providing, that their relative values are fully appreciated and given proper general effect in selecting the value of the propulsive coefficient pc.

Table 3 illustrating the variations in the values of pc accompanying different values of e, e_m and C_r used in conjunction with the values of pc given in Table 2 will be of service in deciding upon the probable value of pc for use with the "Computer."

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DIMENSIONS OF PROPELLERS.

With proper values assigned, in any given case, to the quantities dealt with in the preceding pages, we are then in a position to determine the remaining dimensions and characteristics of the propeller, that are intimately related to each other, consisting of the:—Diameter, Pitch, Revolutions, Slip and Efficiency.

The conditions which surround the design of a ship and machinery, as a general rule rendered it necessary that a value be assigned to one or other of the above, either as a definitely decided figure, or as a limit not to be exceeded. We will now discuss the limitations as are usually found in actual practice.

Diameter—D. Limitations of draft, combined with a suitable immersion of the blades, may place a limit on the diameter of the propeller which cannot be exceeded in a given vessel. With reference to the proper immersion, good practice indicates that the upper part of the disc swept by the blades of the screw should not be nearer the surface of the water than from .1 to .2 of the draft at the stern, always bearing in mind, that the more deeply the screw is immersed, the less will its efficiency be affected by pitching. In vessels of high speed, when information can be had relative to the location and height of the stern wave, propagated by the motion of the vessel, as also change of trim when at the top speed, the above limits of immersion may be modified with reference to the still water line.

Structural considerations, as also the protection of the screws in twin screw ships, may lead to the selection of relatively small diame-

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ters, and, if this can be done without an undue sacrifice of efficiency, it is preferable, in the author's opinion, to that of lapping the screws; this opinion, however, is based on the results of a limited number of applications of this method of affording protection to the propellers, and further experience is needed before definite statements could be made as to the effect on the performance of the screws; meanwhile, decrease in diameter, instead of overlapping, is advised as the safer course to pursue.

Fullness of the afterbody should also be taken into account, along the lines indicated in the discussion on surface ratio.

If no limits are placed on the diameter of the propeller, that obtained from the "Computer," associated with the other co-related dimensions, may be accepted without hesitation as the most suitable. The diameter of propellers for triple screw vessels must depend on a due consideration of all the conditions; but, for the reasons stated under "pitch ratio" discussion, the writer advocates the employment of larger diameter and smaller pitch ratio for the center screw than those of the side propellers.

Pitch—P. The pitch of a propeller may be defined, as the length measured along the axis, which an infinitely small portion of the surface, at a given point on the rear or driving face of the blade, would advance in one revolution, provided, it turned in a solid nut fixed in position. If the pitch increases at successive points from hub to tip of blades, it "increases radially;" if it increases from the leading to the following edges, it "increases axially;" if it is constant at all points of the rear face, it has a "uniform pitch." There is no reliable evidence, as to any benefits arising from the adoption of other than a uniform pitch, in fact, experience has shown generally, that it is rather the

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opposite, and as a result, uniform pitch is becoming more and more, to be standard practice.

Pitch ratio—p. The pitch of a propeller is most conveniently dealt with by relating it to the diameter; the ratio of the pitch to the diameter being known as the pitch ratio, or $P \div D = p$.

Having decided upon a surface ratio suitable for the type of vessel, and with a given diameter or number of revolutions imposed by limit, ing conditions, attendant on the ship or machinery, the "Computer" will give the pitch ratio necessary to adopt for maximum, or such other efficiency of propeller, which a consideration of all the conditions may render desirable to adopt in a given case.

If, on the other hand, to suit a peculiar set of conditions, or, individual preferences, a pitch ratio be definitely decided upon, the most suitable diameter and number of revolutions, associated with the efficiency and other selected characteristics, can be obtained by means of the "Computer."

As already pointed out, in the discussion of surface ratio, vessels with full bluff stern lines require relatively increased diameter, in comparison with that of the propeller suitable for the finer lined ships; this increase may be obtained by the adoption of high pitch ratios, but in these vessels, owing to the unavoidable, closer proximity of the propeller to the hull, high pitch ratio tends to increase the augmentation of resistance, arising from the greater intensity of the suction exerted on the water forward of the propeller, which causes a defect of forward pressure on the stern of the ship. It follows, therefore, that full stern lined vessels should, in general, have smaller surface and lower pitch ratios, than those having medium or fine lines—see values of surface ratio—Table 2.

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While model experiments with propellers indicate that maximum efficiency may be expected over a large range of pitch ratios, say from 1 to 2.5, the conditions ordinarily surrounding the selection of the diameter of propeller and number of revolutions to suit the prevailing types of vessels and their machinery, are such, that in actual practice the pitch ratio is usually found between 1 and 1.6; 1.25 to 1.35 may, therefore, be stated as a good average pitch ratio for all types of vessels, except those with bluff sterns. Pitch ratios below 1.1 are spoken of as low pitch ratios, and the author has obtained good results with pitch ratios as low as .75 and .8 in full cargo ships. The higher pitch ratios are generally most suitable for the smaller types of vessels—small yachts, launches, etc.—but in all cases the final selection will depend on the most suitable co-related diameter, revolutions, and surface ratio.

In the case of triple screw vessels, if the engines are of the same size and power, it will be necessary to assume the same number of revolutions for each of the screws; and, to avoid excessively high slip of center screw, a low pitch ratio should be chosen for that propeller, while the pitch ratio of the side screws should be relatively high; if the maximum efficiency point be maintained with both side and center screws the latter will consequently be larger in diameter. This, the author believes to be the most logical arrangement, not only to obtain the best results when working at full power with the three screws, but also, when using the center propeller only, as it would otherwise be working at an abnormally high and consequently wasteful slip.

Revolutions—R. Ordinarily, each class of vessel calls for the adoption of a certain type of engine as being the most suitable, from either a commercial or technical viewpoint, or, possibly a happy combination of both may decide the choice. In the majority of cases selec-

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tion of type of engine carries with it a given piston speed and length of stroke, thereby fixing the number of revolutions, leaving the diameter pitch and slip ratio for given efficiency, to be determined by aid of the "Computer" after due consideration of the other conditions of the design.

It not infrequently happens, however, that both diameter and revolutions are fixed by the conditions of service, etc., in which case the "Computer" will furnish the pitch ratio corresponding to the best efficiency obtainable.

Slip. If a propeller, fitted behind a ship, worked in an unyielding medium, having characteristics and conditions analagous to that of a solid nut, it would advance at a speed equal to the product of the pitch by the number of revolutions per minute, or other unit of time, and the speed of the ship would necessarily be the same. As water, however, is not an unyielding medium, it is forced back by the action of the propeller with a certain sternward velocity, relatively to a point in the water outside of the influence of the ship and propeller, and the ship moves forward at a speed V, relatively to the same point; the speed V being less than the nominal speed of propeller $P \times R$, by an amount equal to the sternward speed of the water driven back by the propeller, or $P.R - V =$ slip of propeller.

Apparent Slip—S_1. Slip, as ordinarily stated, in trial trip records show the above relationship. For example, take a vessel with a speed of 10 knots, (knot = 6080 feet), driven by a propeller, having a pitch of 16 feet and turning at 76 revolutions per minute. Then:—

Speed of propeller, if it worked in a solid nut $= \frac{76 \times 16 \times 60}{6080} = 12$ kts.

" " ship V $= 10$ "

Consequently, apparent slip of propellor $S_1 = 12 - 10 = 2$ "

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The slip as thus obtained is only the apparent slip S_1 for the reason that the speed or actual advance of the ship is taken relative to a point in the outlying undisturbed water; so far as the performance of the ship is concerned this datum point is quite proper; but, as it is the performance of the propeller that is under consideration, the advance or speed of the ship—and consequently that of the propeller—should be taken with reference to a point in the water in which the propeller is actually working.

Real Slip—S. We have on a previous page discussed the presence of a following current or wake, and the causes of its formation. Due to the presence of this following current or wake the propeller is operating in water, that relatively to the ship, has a speed less than the speed V by an amount equal to the forward speed of the wake. And it is to a point in this water comprising the wake, that the speed or advance of the propeller—and necessarily that of the ship—should be referred. As has been already shown, on page (12) the speed of the water in which the propeller works $= V - wV = V(1-w) = V_1$ where $wV =$ forward speed of wake, stated in terms of the speed of ship V. If, in the example given in preceding paragraph, we assume a forward speed of wake of $wV = 2$ knots, the real slip S is obtained as follows:

Speed of propeller, if it worked in a solid nut $= \frac{76 \times 16 \times 60}{6080} = 12$ kts.

" " vessel minus speed of wake $= V - wV = 10 - 2 = 8$ "

Consequently, real slip of propeller $S = 12 - 8 = 4$ "

Negative Slip—($-S_1$). Occasionally we find negative values of apparent slip S_1 given in trial trip records of screw steamers; this simply indicates in any given case, that the forward speed of wake was greater than the sternward speed of the water driven aft by the pro-

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peller, the latter being the real slip speed; if the speed of wake is equal to the real slip speed, the apparent slip speed will be zero; and with the development of greater wake speeds, true slip remaining constant, the greater will be the negative value of apparent slip.

Negative true slip is an impossibility, as it would mean that a propeller could exert thrust without imparting a sternward velocity to the water on which it acts, which would be an absurdity.

Should negative real slip be indicated by trial results, it may be safely concluded, that an error has been made, either in observing the speed of ship, or number of revolutions, or in the measurement of the pitch of propeller; any one, or more of these sources of error being of request occurrence.

Slip Ratio—s_1 and s. The slip is usually stated as a percentage, or as a ratio, of the speed of propeller. In the examples just given they would be as follows:—

Apparent slip ratio $s_1 = \frac{P.R - V}{P.R} = \frac{2}{12} = .166$ or $16\frac{2}{3}\%$

Real " " $s = \frac{P.R - V_1}{P.R} = \frac{4}{12} = .333$ or $33\frac{1}{3}\%$

If the value of the wake factor w is known in any given case in which the apparent slip ratio has been determined, the real slip ratio may be obtained by solving the equation $s = w + s_1 (1 - w)$

Or, knowing the values of w and real slip ratio, the apparent slip may be obtained by the equation $s_1 = \frac{s - w}{1 - w}$

Consequently, if real slip and apparent slip be known, the wake factor w may be obtained by the equation $w = \frac{s - s_1}{1 - s_1}$

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Real Slip for Maximum Efficiency.—As a result of Froude's experiments, it has been conclusively established, that a propeller of any given type has for every pitch ratio a definite amount of real slip at which it must work to obtain maximum efficiency; a greater or less slip resulting in a lowering of the efficiency.

Table 4 gives the amount of real slip % for different pitch ratios corresponding to maximum efficiency, which is applicable to propellers having elliptical or standard shape of blades, for moderate departures from this shape, the figures are sufficiently accurate for all practical purposes.

The particulars of propellers obtained by the use of the "Computer," implicitly involves the real slip, which can be either computed, or, in cases where maximum efficiency has been used, can be obtained direct from Table 4.

It is recommended by some writers, evidently not familiar with the actual conditions involved in practice, to select the amount of real slip, (which is equivalent to selecting the pitch ratio) and then fix the other dimensions of the propeller to suit; from what has been stated in the preceding pages it may be readily seen that this method, except in special cases, is the least satisfactory procedure; for those who prefer it, however, Table 4 will be of service in furnishing the proper pitch ratio, for maximum efficiency, from which point they can proceed to the determination of the other dimensions of propeller, in the manner best suited to the requirements of the case.

It has been stated in the remarks on pitch ratio, that in actual practice pitch ratios range from 1 to 1.6; from an inspection of Table 4 we find that this corresponds to a working range of real slip, with maximum efficiency, of 18.6% to 22.3%, or an average of 20%.

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From the remarks on the treatment of full lined vessels, in which small pitch ratio was recommended, it follows that these vessels perform best with small slip ratio; this is substantiated by experience with vessels of that type in the majority of trades.

Efficiency.—It is almost unnecessary to say, that in proceeding to the determination of the dimensions of a screw propeller for a given case, maximum efficiency, should be aimed at in the first instance, and should the results, with suitable combinations of surface ratio, pitch ratio, etc., be unsuitable, the efficiency nearest to maximum, which would at the same time give the best fulfillment of all the conditions, should be selected. The efficiency scales on the "Computer" give the decrease in efficiency, which result from the adoption of greater or lesser slip, than that corresponding to the maximum, those to the left of the maximum, on each of the scales, denoting greater slip, and those to the right, lesser slip. Thus, if 69% is accepted as the maximum efficiency obtainable by well designed propellers, and in a given case, if the figure (either to the right or left of the maximum point of the scales of efficiencies *D* or *R*) which is opposite the given pitch ratio, is 2, then the true efficiency of the propeller under consideration would be 69—2 = 67%.

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DIRECTIONS FOR OPERATING THE COMPUTER.

The operating of the "Computer" in solving propeller problems is exceedingly simple, involving only the setting of the several scales according to a certain order, the order depending upon how the problems are stated and the quantities required. The following points, however, should be carefully borne in mind.

Pitch Ratio and Efficiency Scales.—On inspection it will be seen that the upper movable disc has two pitch ratio scales—*D* and *R*. The *D* scale is for use in dealing with the diameter of propellers, and should be operated only in conjunction with the efficiency per cent. scale *D* on the lower movable disc; similarly the *R* scale is for use in dealing with the revolutions, and should be used only in conjunction with the efficiency per cent. scale *R* on the lower disc. The division of each of the efficiency scales, marked "maximum," indicates maximum efficiency of propeller and should be used in all cases where that condition is aimed at; the several divisions of these scales, on each side of the maximum, indicates the loss per cent. in efficiency—from 1 to 10—those to left of the maximum accompanying greater slip, and to the right lesser slip than that belonging to the maximum.

Propulsive Coefficient and Wake per cent. Scales.—The "Computer" is designed with a value of propulsive coeft.: = .5 as standard; when this value is selected as suitable for a given case it is

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unnecessary to operate with the propulsive coefficient scale. Similarly, zero wake was adopted as standard in the design of the wake % scale, for actual values of wake %, the scale is used, in the manner indicated later. Both of these standards are indicated by arrows and are referred to in the succeeding pages as lower and upper arrows respectively.

Revolutions÷Speed (V_1) Scale. In using this scale the value found opposite the selected surface ratio has to be multiplied by V_1 *i. e.*, speed of water in which the propeller works, to obtain the number of revolution; and in starting from a given number of revolutions, these have to be divided by V_1, the resultant figure being picked out on this scale and used in conjunction with the appropriate value of the surface ratio scale. Care must be taken to use V_1 and not speed of ship.

Small Launch Propellers.—The "Computer" can be used for determining the dimensions of propellers having diameters less than two feet, by multiplying the $IHP \times 144$ and using that value on the *IHP* scale; the values of the diameter scale are then to be read as inches instead of feet; and the values of the revolution scale will be equal to $R \div 12\ V_1$ instead of $R \div V_1$; consequently, to obtain the revolutions the values read off the revolution scale are to be multiplied by $12\ V_1$ instead of V_1.

Similarly in starting with given diameter of propeller or number of revolutions, the former should be in inches and the latter divided by 12 and these values, together with $IHP \times 144$ used, in operating the "Computer."

Examples of Operation.—Appended are several examples of problems connected with screw propellers, as are ordinarily met with

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in every day practice: these will serve to illustrate the methods of operation in the several cases.

PRELIMINARY OPERATIONS.—It is to be noted, that in all cases the following preliminary operations are first performed, succeeded by those peculiar to each problem.

r.u.d and r.l.d denote, retaining in position the upper and lower discs respectively.

(1) Selecting values of wake %; surface ratio; and propulsive coefficient from tables, etc.
(2) Setting speed opposite *IHP* (r.l.d).
(3) Bringing upper arrow opposite lower arrow (r.u.d).
(4) Turning lower arrow to wake %, noting V_1 opposite *IHP* (r.l.d) And, where the value of propulsive coeft. is other than the standard = .5.
(5) Bringing upper arrow opposite lower arrow (r.u.d).
(6) Turning selected value of propulsive coeft. to upper arrow (r.l.d).

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EXAMPLES.

Case I. Required Revolutions and Pitch of propeller, at maximum efficiency, for single screw vessel.

Length------270 ft. *IHP*-------1500
Beam---------36 ft. Speed--------12 kts.
Pris: coef----.73 Diameter not to exceed 18.5 ft.

(1) From Table 1. Twin screw division= 9.0
" addition for single screw=11.1
Wake %=20.1
Surface ratio=.43, propulsive coefficient=.5
(2) Set 12 knots opposite 1500 *IHP* (*r.l.d.*)
(3) Bring upper arrow opposite lower arrow(*r.u.d.*)
(4) Turn lower arrow to 20.1% wake. (*r.l.d.*)
Note V_1 opposite 1500 *IHP*=9.6
(a) Set surface ratio .43 opposite 18.5 diameter.
Note pitch ratio opposite max. effcy.=1.11
(b) Set 1.11 pitch ratio opposite max. effcy. (R scale)
Note value $R \div V_1$ opposite .43 surface ratios =6.14
∴ Revolutions=6.14×9.6=58.94
Pitch=18.5×1.11=20.54

Case II. Given Revolutions, required Diameter and Pitch ratio for propellers, at maximum efficiency, for triple screw vessel.

Length------470 ft. *IHP* (one engine)--8000
Beam---------65 ft. Speed ---------- 24 knots
Pris. Coeft.----.58 Revolutions---------- 135

Side Screws.

(1) Wake from Table 1, twin screw division=7%
Surface ratio=.42 propulsive coeft.=.57
(2) Set 24 knots opposite 8000 *IHP* (*r.l.d.*)
(3) Bring upper arrow opposite lower arrow(*r.u.d*)
(4) Turn lower arrow opposite 7% wake (*r.l.d.*)
Note V_1 opposite 8000 *IHP*=22.32
Compute value of $R \div V_1$=135÷22.32=6.05
(5) Bring upper arrow opposite lower arrow(*r.u.d*)
(6) Turn prop.coeft.57 opposite upper arrow(*r.l.d*)
(a) Set surface ratio .42 opposite 6.05 $R \div V_1$ scale
Note Pitch ratio opposite max. effy. (R. Scale)=1.48
(b) Set pitch ratio 1.48 opposite max. effy. (D Scale)
Note Diameter opposite .42 surface ratio= 14.4
∴ Diameter=14.4 and Pitch ratio=1.48

Center Screw.

(1) Wake from Table 1 twin screw division= 7.0
Add for single screw=16.4
Wake %=23 4
Surface ratio=.36 propulsive coefficient=.57
(2) Set 24 knots opposite 8000 *IHP* (*r.l.d.*)
(3) Bring upper arrow opposite lower arrow(*r.u.d.*)
(4) Turn lower arrow opposite 23.4% wake (*r.l.d*)
Note V_1 opposite 8000 *IHP*=18.38
Compute value of $R \div V_1$=135÷18.38=7.34
(5) Bring upper arrow opposite lower arrow(*r.u.d*)
(6) Turn prop.coeft .57opposite upper arrow(*r.l.d*)
(a) Set surf. ratio .36 opposite 7.34 $R \div V$ scale
Note pitch ratio opposite max. effy. (*R* Scale)=1.0
(b) Set pitch ratio 1.0 opposite max. effy. (*D* Scale)
Note Diameter opposite .3 surf. ratio=16.91
∴ Diameter=16.91 and Pitch ratio=1.0

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Case III. Given Pitch ratio, required Diameter and Revolutions for propeller, for single screw ship

Length..... 250	*IHP*.....1000
Beam.....42	Speed.....10
Pris. coeft.....8	Pitch ratio.....8

(1) Wake, Table 1, twin screw division 14.0
add for single screws 10.5
Wake %=24.5
Surface ratio=.3 propulsive coeft.=.5
(2) Set 10 knots opposite 1000 *IHP* (*r.l.d*)
(3) Bring upper arrow opposite lower arrow(*r.u.d.*)
(4) Turn lower arrow opposite 24.5% wake (*r.l.d*)
Note V_1 opposite 1000 *IHP*=7.55
(a) Set pitch ratio .8 opposite max. effy.(*D* Scale)
Note Diameter opposite .3 surf. ratio=20.55
(b) Set pitch ratio .8 opposite max. effy. (*R* scale)
Note value $R \div V_1$ opposite .3 surf. ratio=7.4
Revolutions=7.4×7.55=55.87
∴ Diameter=20.55 and Revs.=55.87

Note:—This is a case where, with maximum efficiency, the diameter would probably be too large for size of vessel, and the revolutions too slow for ordinary type of engine as fitted to this class of vessel: these may be modified by either increasing the surface ratio or accepting a lower efficiency. For example, if 2% loss in efficiency with same surface ratio be accepted, we obtain Diameter=15.5 Revolutions 83.43 and Efficiency =67%.

Case IV. Given Diameter, required Revolutions and Pitch of propeller for Single Screw wooden launch.

Length.....32	Speed 12 miles.....10.4 knots
IHP 15	Diameter.....1 foot 10 inches.

As diameter is less than 2 feet, we have to multiply *IHP*×144 and diameter×12.

IHP 15×144=2160 diameter in inches=22.
(1) Wake, Table 1 twin screw launch 3
Add for single screw 2
Wake %=5
Surface ratio=.47 propulsive coeft.=.45
(2) Set 10.4 knots opposite 2160 *IHP* (*r.l.d.*)
(3) Bring upper arrow opposite lower arrow(*r.u.d.*)
(4) Turn lower arrow opposite 5% wake (*r.l.d.*)
Note V_1 opposite 2160 *IHP*=9.88
(5) Bring upper arrow opposite lower arrow(*r.u.d.*)
(6) Turn prop.coeft .45 opposite upper arrow(*r.l.d*)
(a) Set surf. ratio .47 opposite 22 diameter
Note pitch ratio opposite max. effy. (*D* scale)=1.42
(b) Set pitch ratio 1.42 opposite max. effy.(*R* scale)
Note value $R \div V_1$ opposite .47 surf. ratio= 4.14

As the diameter is less than 2 feet the value of $R \div V_1$ has to be multiplied by $12V_1$
∴ Revolutions=4.14×9.88×12=491
Diameter = 22 inches
Pitch = 22×1.42=30.14 inches

Note:—If revolutions are given the value of $R \div V_1$ is obtained by dividing revs. by $12V_1$ diameter being multiplied by 12 and *IHP* by 144 as above, the diameter would then be read of, in inches, from scale.

A sufficient number of cases have been given in detail to give the reader a good idea of the modus operandi ; problems may be stated in other forms which we will note in brief.

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Case V. Given Diameter and Pitch ratio, required Revolutions and Efficiency. The preliminary operations being the same as already stated in preceding examples.

(a) Set selected surf. ratio to diameter. Note efficiency opposite given pitch ratio.

(b) Set pitch ratio to above noted efficiency on the R efficiency scale, and note value of $R \div V_1$ from which is obtained the number of revolutions in the usual way.

Case VI. Given Revolutions and Pitch ratio, required Diameter and Efficiency.

This case is worked in a similar way to the preceding, only, starting from the $R \div V_1$ scale instead of diameter.

Case VII. Given Diameter and Revolutions, required Pitch ratio and Efficiency, the preliminaries being the same. The value of $R \div V_1$ being computed, the surface ratio is brought opposite the diameter and $R \div V_1$ values, and by inspection of the efficiency and pitch ratio scales, the values which come opposite each other on both D and R scales are the quantities required; a good way is to plot the efficiency values of both R and D scales to an abscissa of pitch ratio, and where they intersect will give the required values of pitch ratio and efficiency.

Froude's Abscissa Values.

The following are the location on the efficiency scales of Froude's abscissa values, and will be of service to those who prefer to use his original estimate of the maximum point of the efficiency surface.

Froude's Abscissa Value.	*Location on Efficiency Scales.*	
16	4.47	To left of maximum with greater slip.
15	3.25	
14	2.26	
13	1.4	
12	.75	
11	.25	
10 = Author's maximum		
9	.13	To right of maximum with lesser slip.
8	.65	
7	1.6	
6	3.25	
5	6.0	

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